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The Association
of
Engineering and Shipbuilding Draughtsmen.

AN INTRODUCTION TO BALL AND ROLLER BEARINGS.

By H. N. STAINES.

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INTRODUCTION.

Although very important components, ball and roller bearings, being small compared with the remainder of a machine, are generally the most neglected during the design stage of the apparatus; consequently, major alterations are very often necessary before the design can be completed.

The purpose of this pamphlet, therefore, is to impart sufficient information to the younger draughtsman or designer to enable him to complete a preliminary layout with the confident knowledge that the bearings selected will be of the correct size and pattern, within reasonable limitations, for the particular duty involved.

Collaboration with the bearing manufacturers before the final design is completed is to be strongly recommended, so as to ensure that the proposed bearing arrangement is the most suitable and economical that can be accommodated within the space available.

In addition to the advantages to be gained by the use of ball and roller bearings for a particular application, it is essential to be acquainted with the various types of bearings in current manufacture and to have some knowledge of the purpose for which they are intended to be used.

Also the practical success of a design depends to a large extent upon the method of mounting being correct, the efficiency of the lubrication and the effectiveness of the protection. All these points will be covered in the following pages, with the hope that useful time will be saved on future designs.

ADVANTAGES.

The substitution of rolling friction for sliding friction presents outstanding advantages to ball and roller bearings over their elder brother, the sleeve or plain pattern bearing.

The chief of these advantages is one of power saving, especially at starting. A considerable proportion of the power lost in good plain bearings is saved by the use of ball and roller bearings and the starting effort can be reduced by as much as 90%.

The reduction in friction also results in reduction in wear, and, in fact, no measurable wear occurs even after many years' service with bearings that have been correctly fitted. This is, of course, of paramount importance to modern machinery, where the maintenance of fine clearances between working parts at high speeds is an essential requirement.

Although the presence of lubricant is necessary for the satisfactory functioning of ball and roller bearings it does not present anything like the difficulty associated with the lubrication of plain bearings, where oil slinging is the rule rather than the exception. In the majority of cases grease may be used to lubricate ball or roller bearings and it possesses the great advantage of being easily retained in the bearing housings. This eliminates the irritating problem of oil slinging. This is of particular importance to industries engaged in the preparation of foodstuffs and to the textile trade, where one spot of oil can ruin many yards of costly fabric. Grease also provides an efficient seal between the shaft and the housing, thus preventing the ingress of dust and moisture. The ease with which grease may be retained in the housing is a great asset from the cost point of view, as one charge of lubricant will last for a considerable period, so that the saving in grease and maintenance costs on large plants is often no mean figure.

Another advantage of particular importance in the design of compact machinery is the saving of space allowed by ball and roller bearings. The width of these bearings is very much less than the shaft diameter as compared with two to four times as great for a plain type bearing.

Finally, the extreme accuracy with which ball and roller bearings are manufactured, is a strong point in their favour, and manufacturers of high class machine tools have been quick to appreciate these points, as better and more uniform work can be produced by machines equipped with precision made bearings.

DESIGN OF BEARINGS.

To the uninitiated the design of ball and roller bearings appears to be a relatively easy matter, but such is not the case ; the mechanics of a bearing are actually of a complex nature.

Having in mind that the machine designer requires the bearings to be accommodated in the smallest possible space, the external dimensions of the bearings have been standardised into ranges, and it is the problem of the bearing designer to obtain the greatest load carrying capacity, coupled with efficiency, within the available dimensions.

In the majority of cases a ball or roller bearing can be divided into four main components :—

(*a*) An inner race.
(*b*) An outer race.
(*c*) Rolling elements.
(*d*) A cage or retainer.

The function of the races is to provide running tracks for the rolling elements, the shape of these tracks varying according to the type of bearing involved, and the duty for which it is intended. These races are manufactured either in direct hardening or case hardening steel, depending upon the size of the bearing or the working conditions.

The rolling elements are generally made from direct hardening steel. The surfaces of these and the race running tracks, should have a hardness figure of not less than Vickers Diamond Hardness No. 800.

After hardening, these components are ground to fine limits of accuracy which are within those specified by the B.S.I. In addition to accuracy special attention is paid to finish to ensure smooth and silent running.

The cage, which separates the rolling elements to prevent them from binding and skidding should be made of a material that has the ability to provide a good bearing surface and, at the same time, combining lightness with strength. From an anti-friction point of view, a copper-alloy has been found to give satisfactory results, and in the case of ball bearings the disadvantage of weight is often minimised by employing cages of pressed metal strip.

Steel strip is also widely used for both ball and roller bearing cages, and another alternative is synthetic resin plastic, which is very popular for ball bearings that have to operate under high speed conditions where the duty is light and the temperature conditions are not abnormal.

Reverting to the races, the design of the running tracks needs careful consideration, for the bearing designer has to accommodate within the external standardised dimensions, the most suitable type and size of rolling element for all kinds of duties.

Whether the tracks have to be suitable for balls, cylindrical rollers, taper rollers or barrel shaped rollers, their shape must be such that frictional resistance is kept to a minimum without surrendering too much load carrying capacity.

In the case of ball journal bearings the contacting areas between the balls and the running tracks are elliptical in shape when the

load is applied to the races, and it is the size of this area of contact that governs the safe load carrying capacity of the bearing. The closer the tracks conform to the shape of the ball, the larger the elliptical area for a given load, and consequently the greater the calculated load carrying capacity.

However, the closer the radius of curvature of the track to the radius of the ball, the greater will be the resistance to rolling, as a result of skidding within the contact area. Therefore, some compromise is necessary and the radius of curvature of the track varies and may be up to $12\frac{1}{2}\%$ greater than the ball radius, depending upon the type of bearing involved and the duty for which it is required.

The greatest radial load carrying capacity would be obtained by allowing zero radial clearance between the balls and their running tracks, as then maximum load distribution would be obtained. However, some allowance has to be made for the fit of the races on their respective seatings. Outer races may contract and inner races expand ; also temperature conditions may still further reduce the radial distance between the tracks. Further, for axial loading a relatively large contact angle is required and therefore some radial clearance is essential in most journal bearings. Reference to this is made later in the booklet.

In the case of cylindrical pattern roller bearings the running tracks are usually parallel. Sometimes it is necessary to resort to tracks having a slightly cambered surface to compensate for malalignment due to shaft deflection or unavoidable inaccuracies in the machine parts, but such tracks suffer the disadvantage of loss in load carrying capacity.

With barrel shaped rollers the tracks are curved accordingly, whilst with tapered rollers the tracks are designed so that the apices of the cones lie at a common point on the axis of rotation of the bearing. With both these types of bearings the frictional resistance is rather greater than with the cylindrical pattern.

The preceding remarks deal very lightly with the design of bearings, and, indeed, are intended to be only elementary as this aspect of ball and roller bearings is a complicated one and only of real importance to the bearing engineer and does not directly concern the machine designer and user who rely upon the bearing manufacturer to provide correctly designed bearings for the work involved.

TYPES OF BEARINGS.

Fig. 1 illustrates some of the more commonly used ball bearings.

At (*a*) is shown the most popular of all ball bearings, viz., the single row, deep groove, rigid pattern no-gap type, which is more widely used than any other. This type of bearing was originally

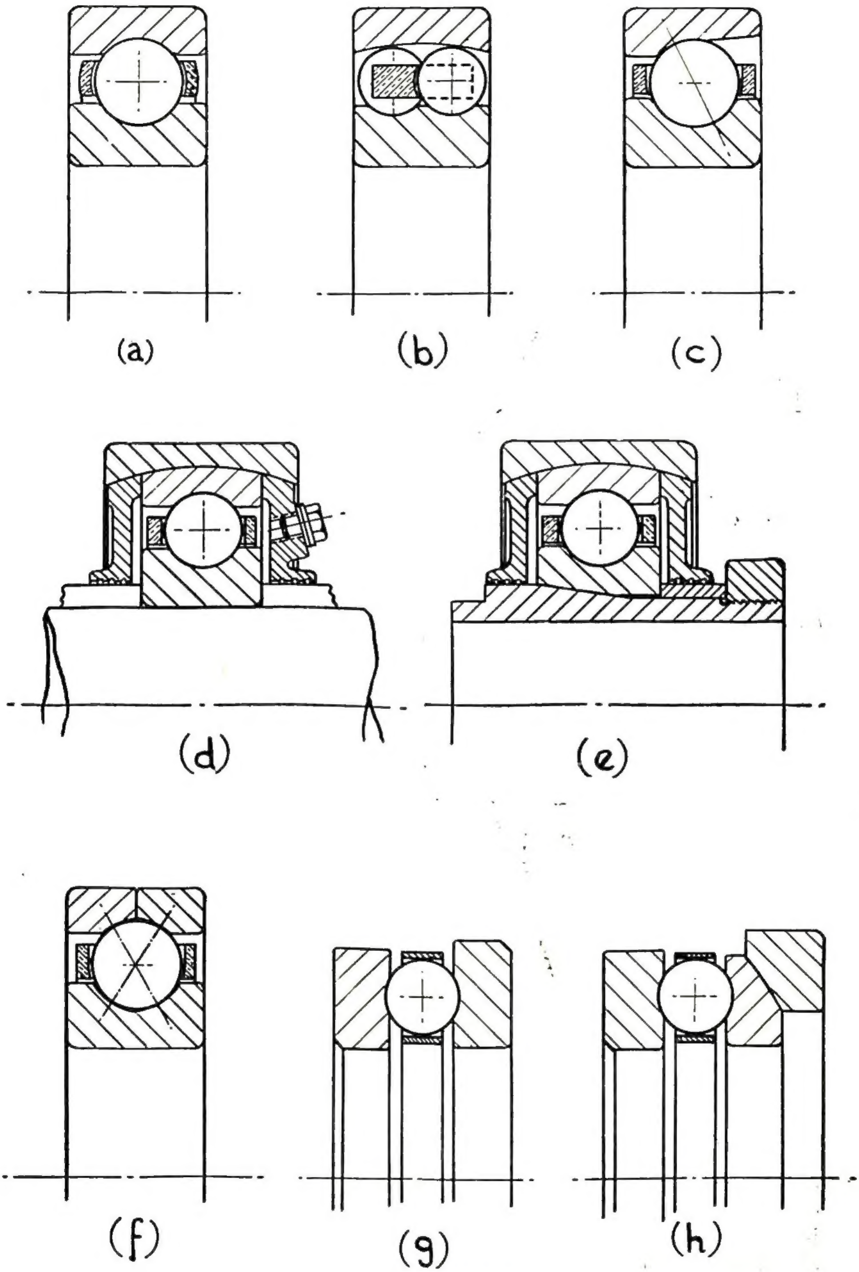


Fig. 1—Types of Ball Bearings.

intended for carrying radial loads only, but for many years it has been extensively used for dealing with limited axial loads or combined radial and axial loading, with every success. One point to be borne in mind with this pattern bearing is that under a reversing thrust load it allows appreciable end play.

This end play is due to two essential features of the bearing, viz., the radius of curvature of the grooved track is slightly greater than the radius of the ball and the other is the internal clearance between the balls and running tracks, known as diametral clearance.

At this juncture a few remarks clarifying the subject of diametral clearance will, perhaps, be opportune, as it is one which is frequently misunderstood.

All radial bearings, both ball and roller pattern, are normally manufactured with three ranges of diametral clearance, the most suitable range to employ depending upon the particular application involved.

The various manufacturers have adopted their own methods of indicating the grade of fit but the more common are one, two, and three dots, or C2, Standard and C3 respectively. For the sake of simplicity, it is proposed to confine the remarks to the former designations, which are identified by the appropriate number of polished circles on the side of one of the bearing races. These markings have no significance other than identifying the grade of slackness. They do not, as is assumed by some engineers, indicate the quality of the bearings.

One dot ('o') fit bearings have the minimum amount of slackness and are only used where the axial and/or radial play must be kept to a minimum. When bearings with this slackness are employed, special care must be taken to avoid expanding the races by making them too tight a fit on their seatings and they should not be used where heat is liable to be transmitted to the bearings through the inner races. Under such conditions the amount of expansion of the inner race is higher than that of the outer race, with the consequence that the diametral clearance is reduced under working conditions and there is danger of all the slackness being eliminated.

Two dot ('oo') fit bearings have the intermediate range of clearance and the greater slackness therefore permits more latitude in the fit of the races on their seatings. Two dot is the standard fit for ball journal bearings used in general engineering applications.

Three dot ('ooo') fit bearings have a larger amount of diametral clearance than 'oo' and should be used where both the inner and outer races are made an interference fit, or where only one race is an interference fit, but there is likely to be some further loss of clearance due to temperature difference between the races.

Three dot is the standard fit for roller bearings on general engineering work.

It is not difficult to appreciate that any increase in diametral slackness in a ball journal bearing, will result in increased axial play which, at the same time, increases the possible angle of contact between the balls and their running tracks (see Fig. 2).

The ability of a ball bearing to withstand axial loading depends on the contact angle and therefore when ball journal bearings are needed to carry thrust load the three dot fit should be used.

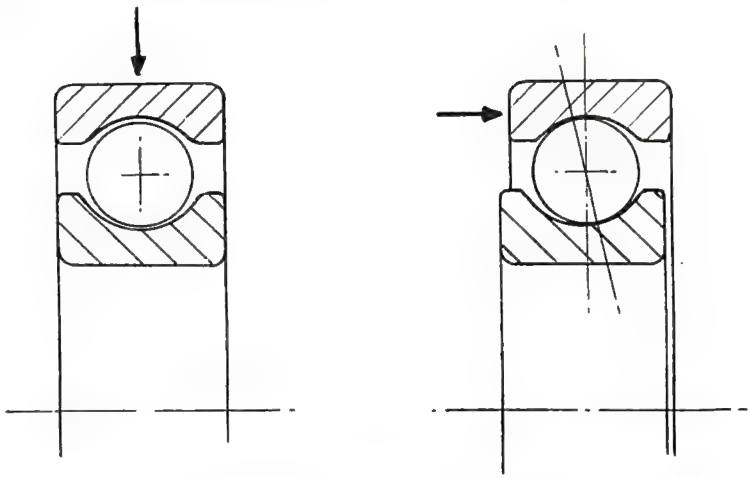
A modification of the standard ball journal bearing is the Angular Contact Pattern, as shewn at illustration (*c*) on Fig. 1.

As the name implies, the line connecting the points of contact between the balls and their running tracks, does not lie in a direction perpendicular to the axis of the shaft. Also, the outer race is open sided to enable a larger number of balls to be fitted than in the ball journal type and as a result of these variations, the bearing is able to carry a substantial amount of axial loading in addition to radial loads.

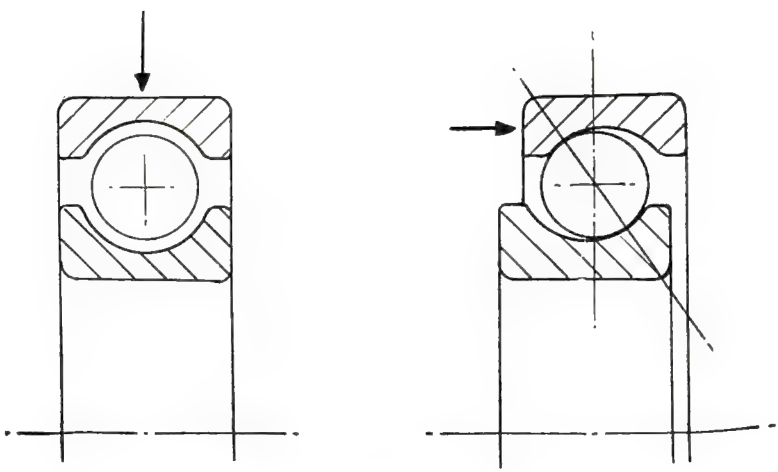
It is usual for the bearings to be mounted in pairs and adjusted against one another, so as to eliminate the initial slackness in the bearings and to maintain the balls in their correct position on the running tracks. When fitted in this manner they are suitable for carrying radial and axial loads in any proportion. When mounted singly, however, it is essential that the axial load at all times exceeds the radial load and is constant in direction.

A further modification to the ball journal bearing is the Duplex bearing as shewn at illustration (*f*) on Fig. 1. This bearing has two alternative points of support for the balls, on both the inner and outer running tracks, enabling axial load to be carried in either direction. This necessitates a certain amount of slackness to be present in the bearing under running conditions which gives rise to a small amount of end play and therefore the bearing is not adjustable. Duplex bearings can be used for pure thrust load in either direction, or thrust load combined with journal load, but it is important that if journal loading is present it must be of appreciably smaller magnitude than the thrust loading, as otherwise the balls may contact the tracks at three or possibly four points, resulting in skidding, heating and consequently, wear. The illustration shows a bearing with a two-piece outer race and a one-piece inner race, but duplex bearings are also manufactured with a one-piece outer race and a two-piece inner race, the choice of pattern to employ depending upon the conditions of application.

A further type of ball bearing is the thrust washer pattern, as shewn at illustration (*g*) of Fig. 1. It will be perceived that it



BEARING WITH NORMAL SLACKNESS.



BEARING WITH INCREASED SLACKNESS.

Fig. 2.

is in effect an angular contact bearing with a contact angle of 90° and it is not suitable for carrying radial loads. This type of bearing is more susceptible to the effects of centrifugal force, acting on the balls, than the angular contact pattern and in these days when speeds of rotating parts become increasingly higher, the thrust washer bearing is being largely superseded by the angular contact pattern.

However, for applications involving heavy and/or shock loading at slow speed, such as grinding pan footsteps, railway turntables, extruding machines and the like, it is often the most suitable type of bearing to use. To obtain the maximum load carrying capacity it is necessary that all the balls are evenly loaded and therefore the abutments for the races of these bearings must be flat and square with the axis of rotation.

So far the only bearings dealt with have been rigid pattern but there is another series, known as self-aligning bearings, which have to be employed when it is not practical to obtain and maintain accurate alignment between the bearing race seatings.

An inexpensive self-aligning ball bearing is the double row pattern, as shewn by illustration (*b*) on Fig. 1. In this type of bearing the self-aligning feature is provided by the spherical track in the outer race and as this is incorporated within the bearing the external dimensions are the same as the corresponding size of rigid bearing. However, due to the comparatively flat outer track which does, of course, give less support to the balls than that afforded by the comparatively close curvature of the deep groove pattern, its load carrying capacity is lower.

Also, as a result of the small angle of contact between the outer race track and the balls it is only intended for positions where there is no appreciable axial loading. Illustrations (*d*) and (*e*) show self-aligning ball bearings where the self-aligning feature is provided by spherical surfaces external to the working components of the bearing. These take up a little more space than the corresponding rigid bearing but they have the same load carrying capacity. Also the end covers fitted to the bearings swivel with the races, thus giving the advantage of maintaining the correct clearance with the rotating parts and providing constant efficient protection under normal conditions. As will be seen, type (*d*) seats directly on the shaft, whereas type (*e*) is provided with an adapter sleeve and nut which enables it to be fitted to commercial shafting without having to machine the seating to fine limits of accuracy. Illustration (*h*) shows a thrust bearing fitted with a self-aligning seating ring.

When, for a given shaft diameter, the load is beyond the carrying capacity of a ball bearing, or shock loading is involved, or there

is a possibility of severe unknown loads, then a roller bearing has to be considered and Fig. 3 illustrates some of the common types that are available.

Illustration (*j*) shows the standard pattern, which consists of a plain annular outer race and a channelled or lipped inner race. It will be appreciated that this pattern bearing will carry radial load only and cannot locate the parts endwise. Roller bearings of this type have a load carrying capacity considerably higher than a ball journal bearing of the same external dimensions. These cylindrical roller bearings are manufactured with various lip formations, as shewn in illustrations (*k*) to (*s*), and have been developed to suit the varying mounting requirements, some of which will be described later. It should be mentioned that where lips are provided on both races, the bearings are capable of dealing with location duty and even carrying light and intermittent axial loads.

Illustration (*t*) shows another member of the roller bearing family, viz., the needle roller bearing. This comprises an inner and outer race ; the essential character of this bearing being that it has a full row of long, small diameter rollers, known as needle rollers. There is, in consequence, no retaining cage, and the clearance between the rollers is kept to a minimum in order to avoid any tendency for the rollers to skew. Owing to the proportions of the rollers, needle bearings are not suitable for dealing with axial loads or even for location duty, and one of the races is therefore made plain without lips. The roller being rather long compared with its diameter, renders the bearings very susceptible to the effects of malalignment and they are therefore not intended to supersede the normal pattern roller bearing, except in applications where space, weight and other conditions render this necessary.

The needle roller bearing was first developed for use in the connecting rods of internal combustion engines, but it has since been successfully used in numerous applications where there is oscillating movement, or where there is fluctuation of speed or reversal of load such as cam rollers, valve rocker arms, aircraft controls, etc.

A different pattern from the cylindrical pattern roller bearing is the taper roller type as shown in illustration (*u*). In this type of bearing, as previously mentioned, the running tracks and rolling elements are conical in shape and should be accurately constructed so that the apices of the cone angles meet at a common point on an extended axis of the bearing. This bearing is really the roller equivalent of the angular contact pattern ball bearing inasmuch as it will carry both radial and axial loading.

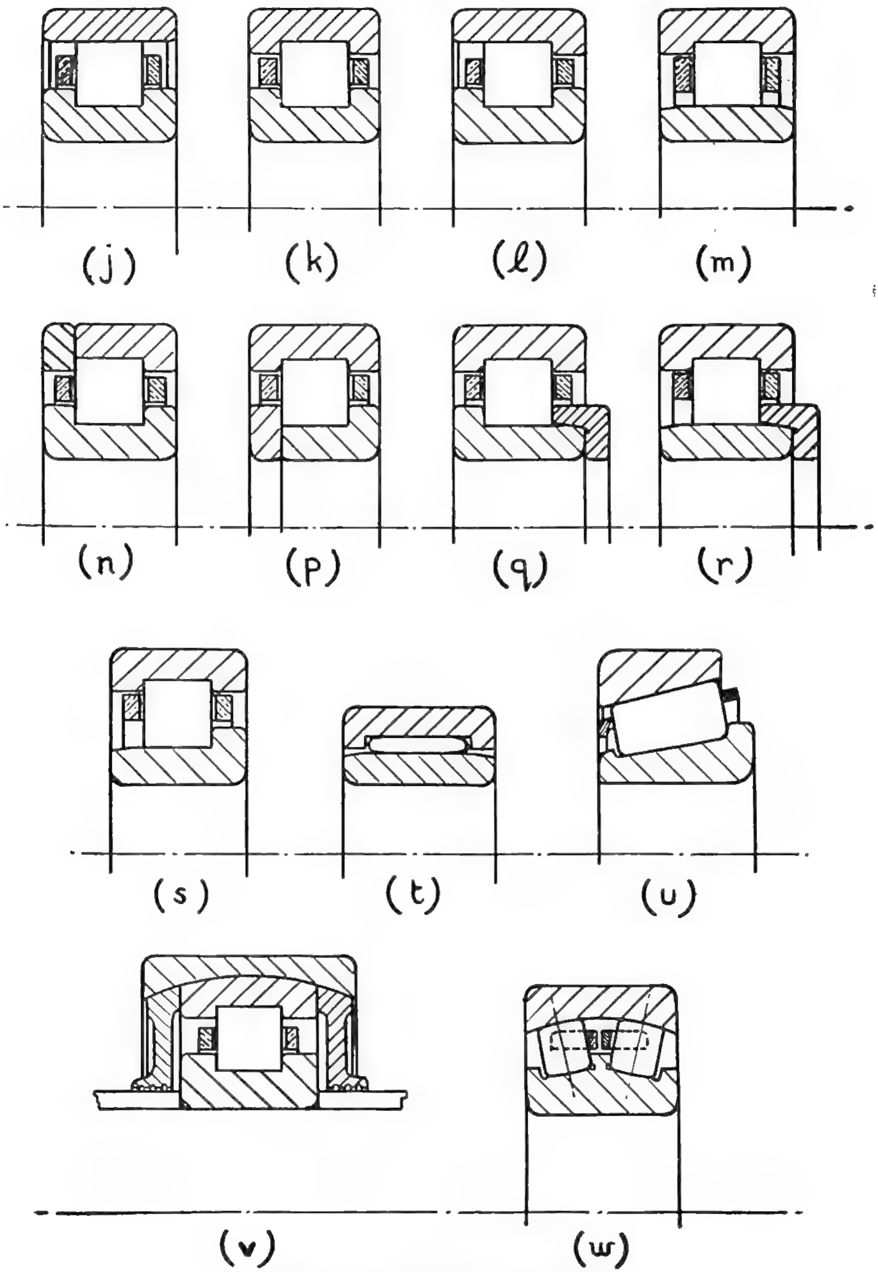


Fig. 3—Types of Roller Bearings.

As with ball bearings, roller bearings are also manufactured with self-aligning features, one type being shown in illustration (*v*) which can also be obtained in the adapter sleeve pattern, similar to Fig. 1 illustration (*e*). A further pattern of self-aligning roller bearing is shown in illustration (*w*) which is similar to the double row self-aligning ball bearing except that the balls are replaced by barrel-shaped rollers, providing a considerable increase in load carrying capacity.

SELECTION OF BEARINGS.

Having acquired some knowledge of the types of bearings that can be obtained and the forms of loading for which they are intended, it is now necessary to determine what size of bearing will be required for any particular application involved. At this stage it is as well to remember that the knowledge required to accomplish this can only be acquired by long practice, backed up by the recorded results of earlier applications. However, provided the loads imposed on the bearings are correctly calculated, then it should be possible to make a reasonable selection of bearing sizes by following the instructions regarding factors of safety which are published in most of the well-known bearing manufacturers' catalogues. Actually, it is very often not at all easy to evaluate the magnitude of the load, especially in the cases where shock components are involved and it is in these cases that experience is the only safe guide and it is only fair to repeat that, when in doubt, collaboration with the bearing manufacturer is to be strongly recommended.

METHODS OF MOUNTING.

The success or failure of a ball or roller bearing arrangement depends to a great extent upon the accuracy with which they are mounted and there are a few elementary rules, attention to which will be amply rewarded by extended trouble-free service. In order to prevent creep of the races it is important to ensure that all revolving races of bearings, dealing with radial loads are made an interference fit on their seatings.

'Creep' is the name applied to the slow rotation of a race relative to its seating and is due entirely to the direction of the radial load constantly changing relative to the race, which has been incorrectly made a loose or a push fit on its seating. It is really a combined sliding and rolling action which will rapidly destroy the surface of the seating if allowed to continue.

Also, of course, the freed metallic particles due to the wear will damage the running tracks and the rolling elements.

If the load is constant in direction then there is no tendency for stationary races to creep and therefore such races may be made a sliding or push fit on their seatings. If, however, out of balance loading, of sufficient magnitude, is present, then the stationary races must also be made an interference fit on their seatings. The reason for this will be readily appreciated when it is explained that the force, due to an unbalanced load, constantly rotates around the stationary race, giving rise to the conditions which result in creep. Sometimes a user will resort to the use of keys and keyways to prevent creep, but this method is to be deplored as such devices quickly wear under the constant chafing and the only safe method is to make the race of a radial bearing an interference fit.

The abutment faces, against which the races should be firmly pressed, must be machined true and square with the axis of rotation in order to avoid canting of the races. All inner race seatings and housing bores must be truly cylindrical so as to avoid danger of distorting the races and consequently the running tracks.

It is very important to ensure that no single length of shaft is located in an axial direction by more than one bearing in each direction. Failure to do this can result in imposing severe axial preload on the bearings with subsequent breakdown of the running tracks and rolling elements. It is usual practice to locate the rotating parts endways in both directions by the same bearing, but where the assembly would be facilitated by controlling the parts endways by two bearings, it is permissible for one bearing to locate the parts in one direction only and the other bearing to locate in the reverse direction.

During assembly it is essential that absolute cleanliness is observed as even the most elaborate of protective devices will be of no value whatsoever if the bearing being fitted contains foreign matter. It is important, therefore, not to remove the bearings from their protective wrappings until they are actually required.

One of the most common and simple methods of mounting consists of two ball journal pattern bearings on the lines of Fig. 4. This illustrates a horizontal rotating shaft on which the bearing inner races should be made an interference fit and clamped firmly endways. The outer races, being stationary members, may be made a push fit in the bores of the housings and one of these races should be held endways, so that this bearing will locate the shaft in both directions. The outer race of the non-locating bearing must therefore be left clear endways so that it may align itself opposite the inner race without imposing a permanent end thrust on both bearings. The end covers, enclosing the bearings, should be made a fine running clearance where they approach the shaft and should be spigoted into the housing bore to ensure concentricity.

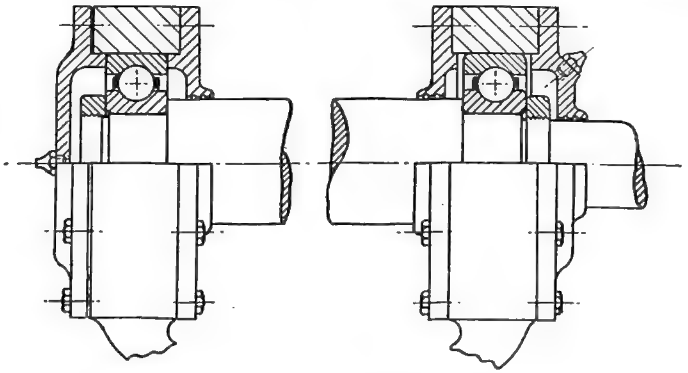


Fig. 4—Ball Bearings to Horizontal Shaft

Fig. 5 also shows a horizontal shaft mounted on two ball bearings but in this case each bearing locates the shaft in one direction. The outer races of both bearings must be a good push fit on their seatings and a small but definite clearance must be provided between the faces of the outer races and the adjacent end cover spigots. This end clearance must be sufficient to allow for relative axial expansion between the shaft and housing and also to accommodate cumulative tolerances of machined parts. Paying attention to this point ensures elimination of possible trouble due to permanent severe axial pre-load. This arrangement is quite suitable for applications where it is impracticable to provide a clamping shoulder for one of the outer races and where appreciable end play is not detrimental to the satisfactory functioning of the machine.

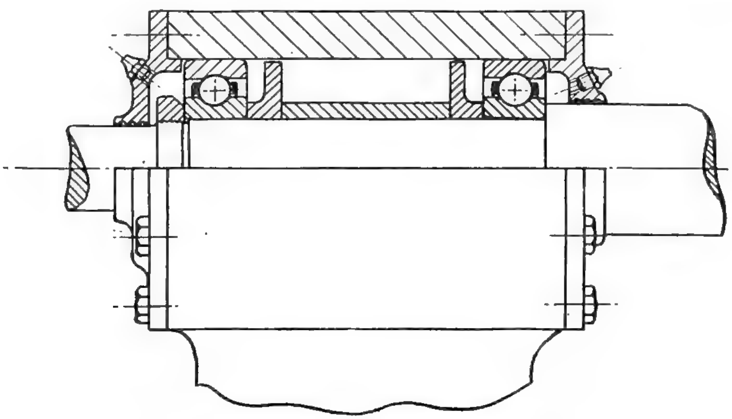


Fig. 5—Ball Bearings to Horizontal Shaft.

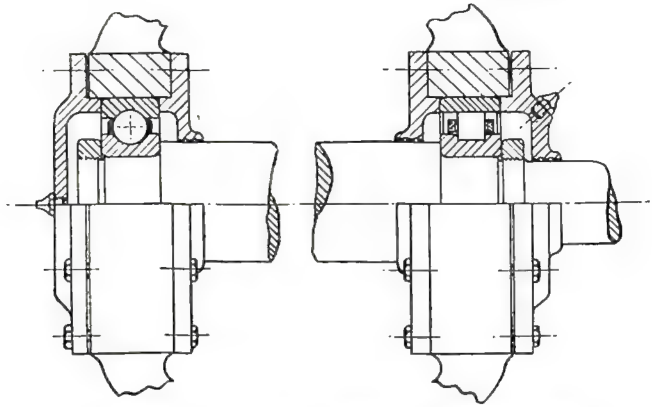


Fig. 6—Ball and Roller Bearings to Electric Motor.

A common arrangement employing one ball and one roller bearing is shown on Fig. 6. The application illustrated is the armature shaft of an electric motor and all the inner and outer races are held endways on their respective seatings, the shaft being located axially by the ball bearing. The inner races are, of course, rotating members and must therefore be made an interference fit on their seatings. The outer races, although stationary members, should be made slightly tighter than a push fit to avoid danger of creep due to possible out-of-balance loading.

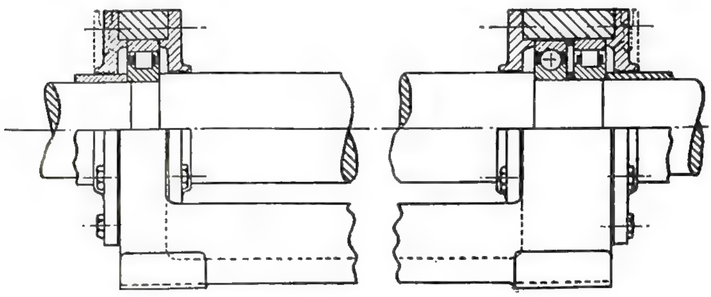


Fig. 7—Ball and Roller Bearings to Fan Shaft.

Fig. 7 illustrates a mounting incorporating two roller journal bearings and a ball location bearing, the application in this instance being to a fan shaft. With this arrangement the roller bearings at each end carry the radial loads, whilst the axial load is taken care of by the ball location bearing. It should be mentioned that these ball location bearings are similar in construction to the ordinary ball journal bearings but they are made slightly

smaller than standard on the outside diameter to ensure they carry no radial loading when fitted in housings bored to suit standard roller bearings. Where the speed is high the ball location bearing should be spaced slightly apart from the adjacent roller bearing by means of distance pieces between the inner and outer races, to allow sufficient grease space between the bearings to ensure satisfactory lubrication of the moving elements.

When the axial load is heavier than may safely be dealt with by a ball location bearing, a duplex pattern bearing can be employed instead. The duplex bearing has a higher thrust load carrying capacity than a ball location bearing of equivalent size and has the advantage that the method of mounting is unchanged.

Where it is desired that the axial play shall be reduced to a minimum, an arrangement comprising the angular contact pattern bearing may usually be used. The bearings must be fitted in pairs and adjusted endwise against one another to eliminate the end play in the shaft.

Fig. 8 illustrates one method of applying angular contact type bearings : in this case the application being to the rear axle of a motor car. It will be seen that both the bevel pinion shaft and the differential are mounted on angular contact bearings which provide both a very simple and inexpensive arrangement.

The bearings are fitted with their open sides towards one another and endwise adjustment is carried out in the case of the pinion shaft by means of shims interposed between the end cover flange and the face of the housing, whilst for the differential bearings it is effected by screwed end covers.

Fig. 9 shows the double row self-aligning pattern ball bearing fitted to the return idler of a belt conveyor and clearly illustrates how this type of bearing can be simply applied to applications of this nature, where the loading conditions are not severe.

For more arduous loading conditions where a self-aligning arrangement is necessary the self-aligning pattern roller bearing is often employed, on the lines of the arrangement shown in Fig. 10. The application in this instance is a fan shaft where the conditions are reasonably clean and therefore the self-aligning end covers, incorporated in the bearings are ideally suitable for retaining the grease in the bearings and preventing the ingress of any light foreign matter that may be present.

At the position opposite to the fan runner one of the self-aligning end covers has been omitted, as it was possible to utilise a blind cover at this end. The method of mounting is self-explanatory and need not be elaborated upon, but it should be mentioned that with this type of bearing it is important that the rotating parts

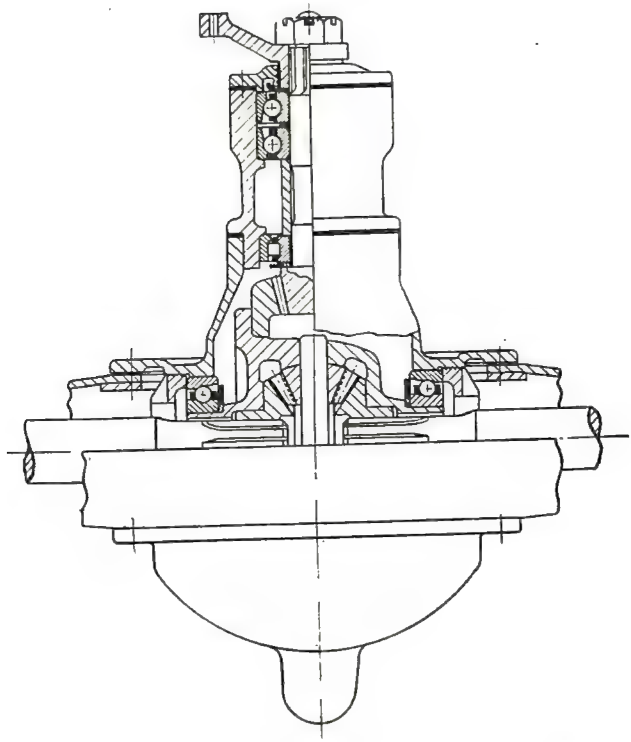


Fig. 8—Ball and Roller Bearings to Rear Axle Drive.

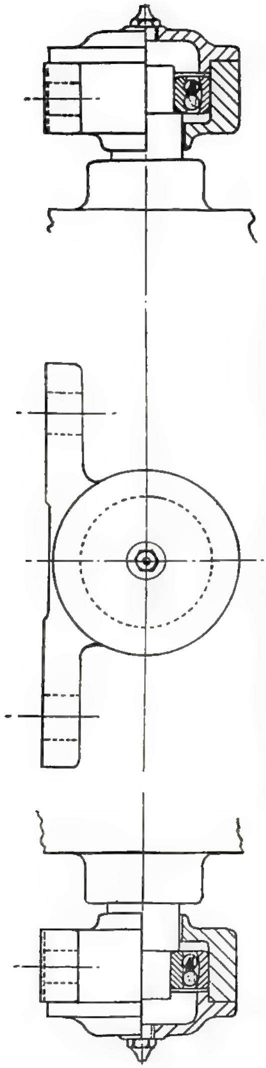


Fig. 9—Self-Aligning Ball Bearings to Conveyor Roller.

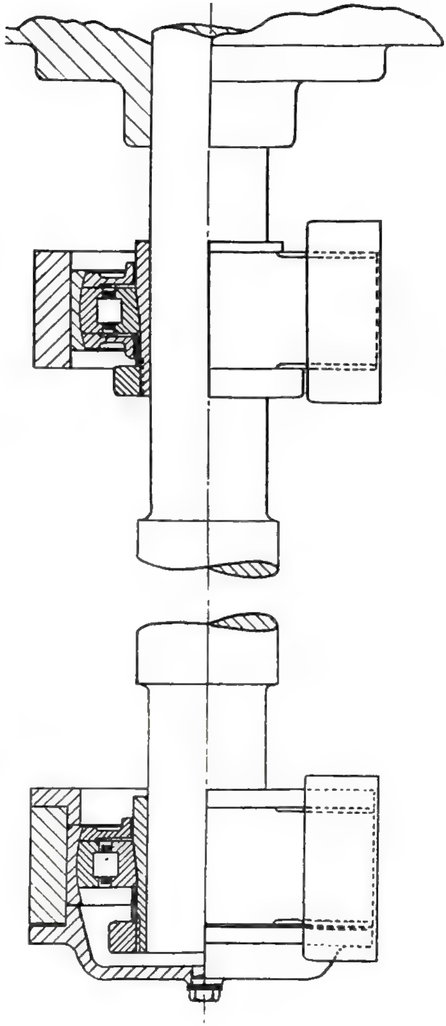


Fig. 10—Self-Aligning Roller Bearings to Fan Shaft.

are in good dynamic balance to avoid the danger of creep, mentioned earlier in the pamphlet.

From an inspection of the internal construction of this type of bearing it will be appreciated that free swivelling of the outer race could not occur if the mating surfaces were a tight fit and therefore any out-of-balance loading would result in relative movement between these surfaces.

Fig. 11 illustrates two examples of employing needle roller bearings, the upper sketch showing one of the links used in a locomotive valve gear, whilst the lower sketch illustrates the gudgeon pin of an internal combustion engine. In the case of the valve gear complete bearings are shown, and they are of the pattern which incorporates retaining pieces so that assembly and dis-assembly is greatly facilitated, because the inner and outer races can be separated without danger of losing the needle rollers. For the gudgeon pin the needles run directly on the pin, in order to save space, and in this case the pin would have to be hardened and accurately ground by the manufacturers of this component.

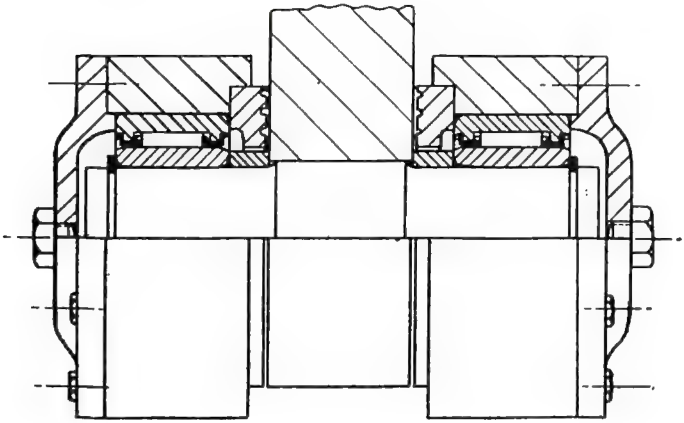
It will be noticed that the space taken up by these bearings is small when compared with the normal pattern roller bearing.

When considering the application of ball and roller bearings during the design stage of a machine some thought must be given to the question of disassembly.

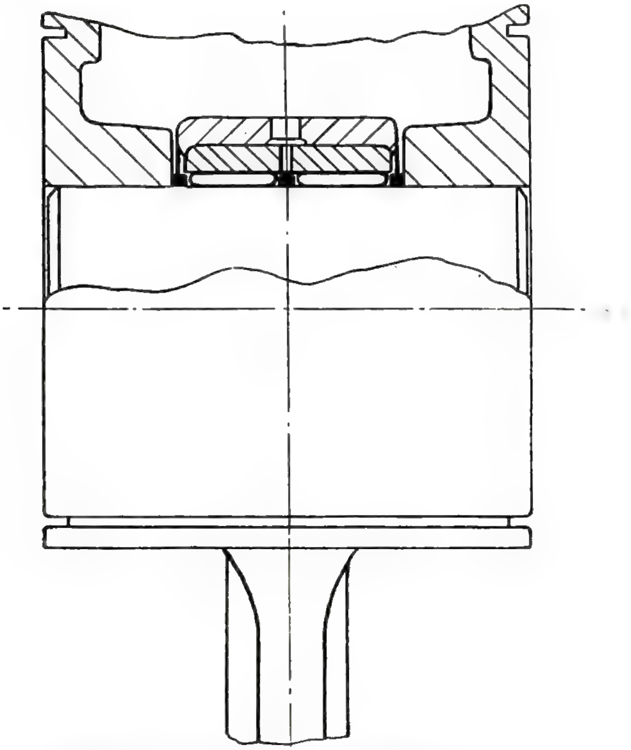
Not only should the parts be readily assembled but they should also be easily dismantled if this should be necessary for maintenance purposes. It often happens that perfectly good bearings have to be destroyed during disassembly of a machine because due care was not given to this point by the designer.

A good example of foresight in this respect is shown in Fig. 12, which illustrates the arrangement which has been virtually standardised on railway traction motors. A very straightforward design has been obtained by using cylindrical roller bearings with two of the special lip arrangements illustrated in Fig. 3. It will be readily appreciated that the end shields of the motor may be withdrawn complete with the bearing outer races, cages and rollers and it is unnecessary to remove either the inner or outer races from their seatings.

Another application in which ease of assembly and dismantling has to be considered is shown in Fig. 13, which illustrates a popular design of axle-box used on industrial and steel works cars. In this, as in the case of the traction motor, end location is taken care of by lipped pattern roller bearings and it is interesting to note that in respect to traction motors these bearings have proved to



Needle Roller Bearings to Locomotive Valve Gear.



Needle Roller Bearings to Gudgeon Pin.

Fig. 11.

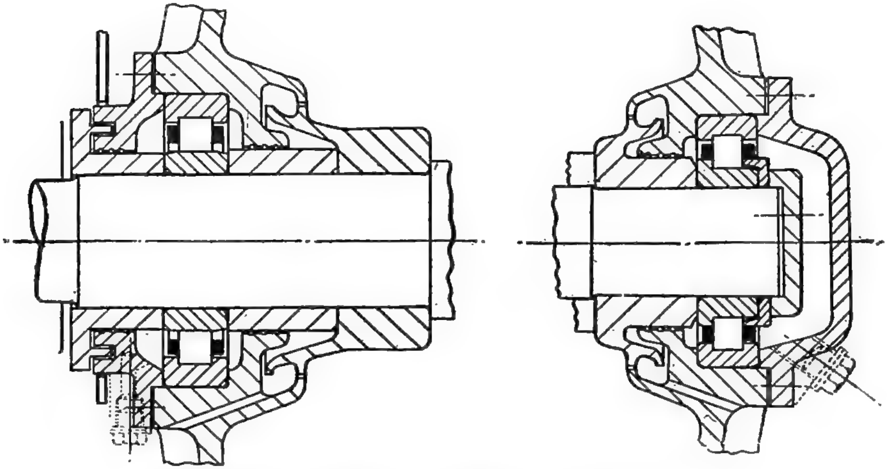


Fig. 12—Roller Bearings to Railway Traction Motor.

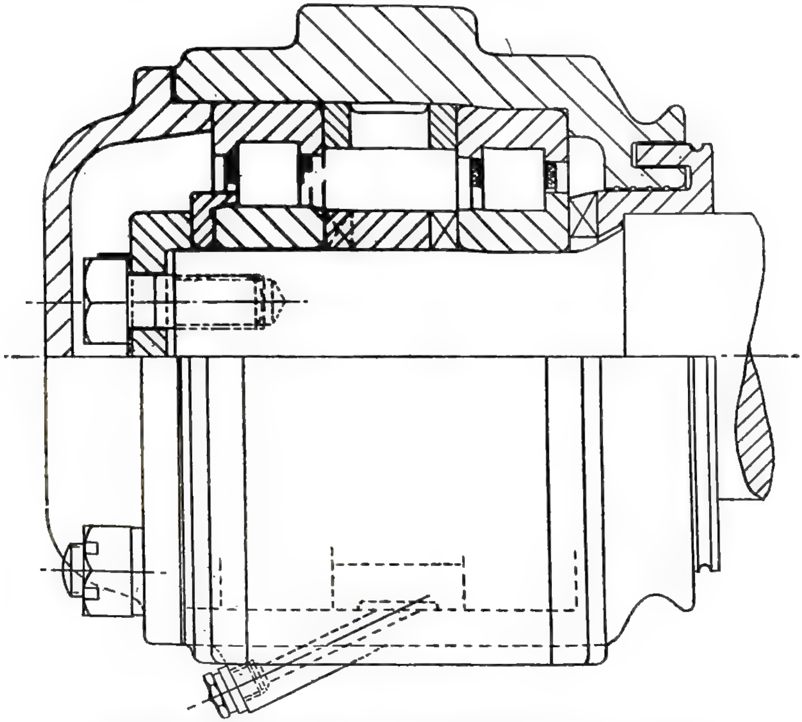


Fig. 13—Roller Bearings to Steel Works Car Axle Box.

be highly satisfactory when dealing with light end thrust set up by spiral gears.

Sometimes it is inconvenient or even impracticable to provide clamping faces for the outer race of a ball bearing which is required to locate the rotating parts axially. This obstacle is very often encountered in automobile gearboxes and in order to help users

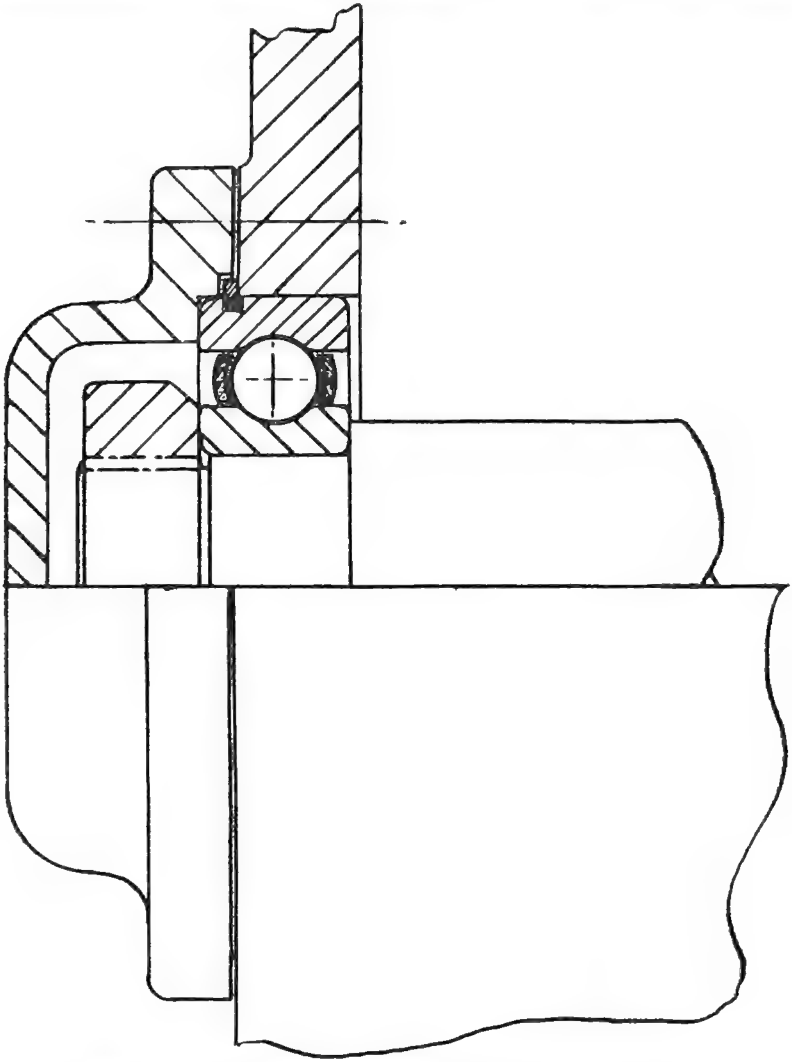


Fig. 14—Method of Mounting for "Snap-Ring" Pattern Bearing.

to overcome this difficulty most bearing manufacturers now supply a limited range of bearings having a groove in the outside diameter of the outer race. This groove will accommodate a split ring which will permit the omission of one abutment face in the bore of the housing, as will be readily appreciated by referring to Fig. 14 which illustrates the method of mounting for the "snap-ring" pattern bearing. The main thrust load is taken against the abutment face provided by the end cover, whilst any backlash or light axial reverse load is taken through the split ring.

The principles of mounting involved for vertical shafts are, in general, the same as for horizontal shafts but special provision must be made for retaining the lubricant.

Fig. 15 shows a typical arrangement which is commonly employed for medium size vertical motors and it will be noticed that internal throwers are positioned below the bearings and are designed to form labyrinth seals in conjunction with internal lips on the lower covers of the housings. Where heavier reversing axial loads have to be catered for an arrangement as in Fig. 16 can usually be employed which incorporates two angular contact pattern bearings adjusted together.

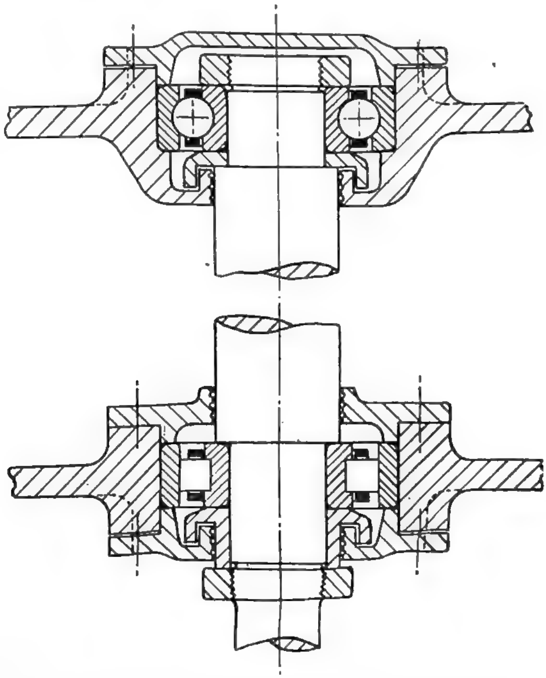


Fig. 15—Ball and Roller Bearings to Vertical Motor.

Fig. 17 illustrates one of the heavy applications, namely, a locomotive turntable. Ball thrust bearings applied to these turntables have proved a successful commercial proposition, as due to the inaccessibility of the pivot constant maintenance is a virtual impossibility. With a bearing as shown it is only necessary to re-charge the bearing housing with grease annually.

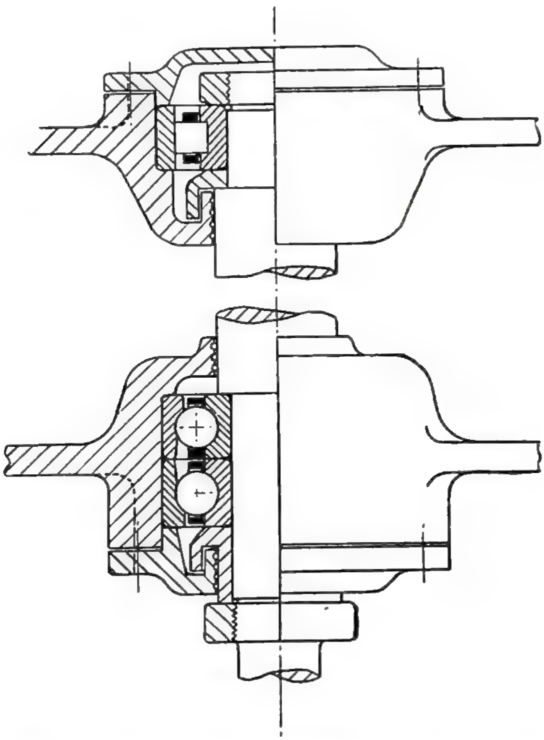


Fig. 16—Ball and Roller Bearings to Vertical Shaft.

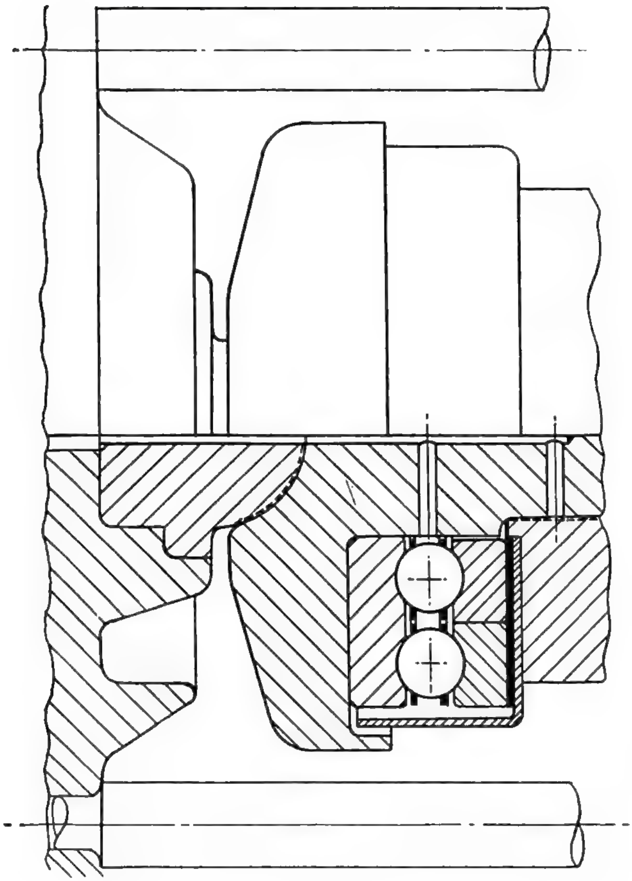


Fig. 17—Ball Thrust Bearing to Locomotive Turntable.

PROTECTION.

The majority of the illustrations have, up to this stage, shown protective covers suitable for reasonably clean conditions but there are many applications where additional protection is necessary and this can best be appreciated by illustrated examples of typical closures.

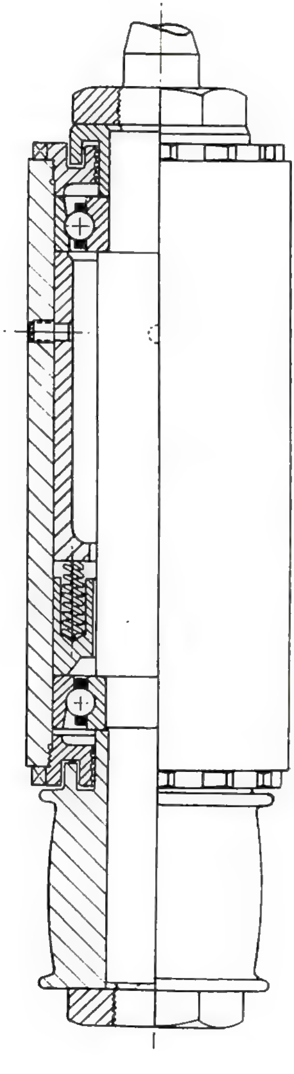


Fig. 18—Ball Bearings to Internal Grinding Spindle.

It cannot be emphasised too strongly that even the highest quality ball or roller bearing will rapidly fail if foreign matter is allowed access to the running tracks and rolling elements. A good example of adverse conditions are those encountered during the grinding of metals, and Fig. 18 shows the form of labyrinth which has been successfully adopted on grinding spindle applications.

A word or two concerning the bearing arrangement shown may be of interest as it is a mounting which has proved to be very successful. The angular contact pattern bearings involved are spring loaded in order to maintain a constant pressure on the balls, which ensures they are always running in their correct positions on the tracks and are therefore unaffected by any relative expansion that may occur between the spindle and the housing. The bearings used on machine tool spindles are normally supplied to extra fine limits of accuracy to ensure the utmost true running of the spindle.

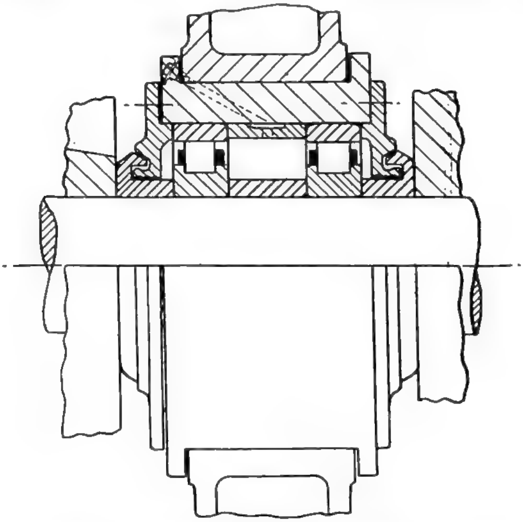


Fig. 19—Method of Protecting Roller Bearings on Grinding Roll.

Fig. 19 shows the applications of roller bearings to a grinding roll where moisture as well as foreign matter is present. In this case the rotating throwers are of conical form and gutters are formed in the end covers to catch any water that may have penetrated past the outer thrower, and thus prevent its obtaining direct access to the joint.

With applications such as paper making machinery, where there are large quantities of water present, really adequate protection

is essential and Fig. 20 illustrates the form of labyrinth employed on breast rolls. Although this may appear to be complicated, it is really efficient and its initial cost far outweighs the maintenance costs that would occur if a more simple labyrinth were used.

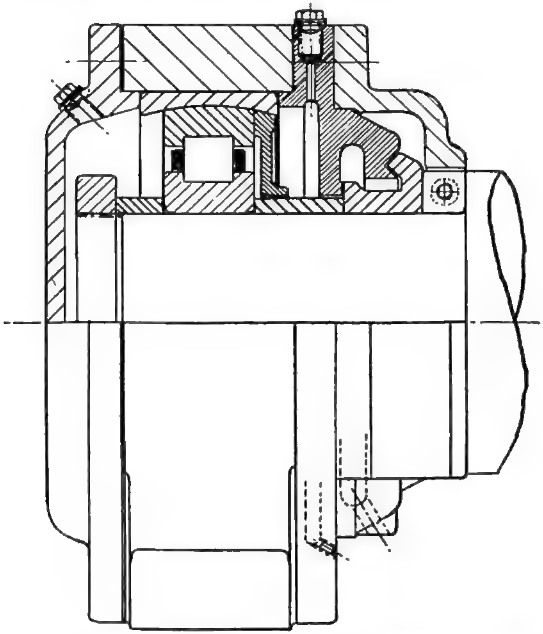


Fig. 20—Method of Protecting Roller Bearings on Breast Roll.

Sometimes it is neither convenient nor practicable to use additional end covers, and Fig. 21 shows how simple sheet metal shrouds can be used to protect a standard plummer block pattern bearing when employed in a dirty atmosphere. In this case, of course, the bearing is fitted with its own self-aligning end covers.

Fig. 22 illustrates how labyrinth protection may be applied to a double row self-aligning ball bearing without impairing the self-aligning feature. The application involved was an eccentric and it will be seen that the end covers and rotating throwers are provided with spherical surfaces, the radii of which are struck from the same centre as that of the spherical track in the outer race.

For automobile applications the modern tendency is to employ proprietary seals, on the lines of that shown in Fig. 23. These are quite effective in excluding dirt and moisture provided the speed is not excessive and it can be ensured that the lubricant reaches the washers to prevent their hardening and damaging the shaft.

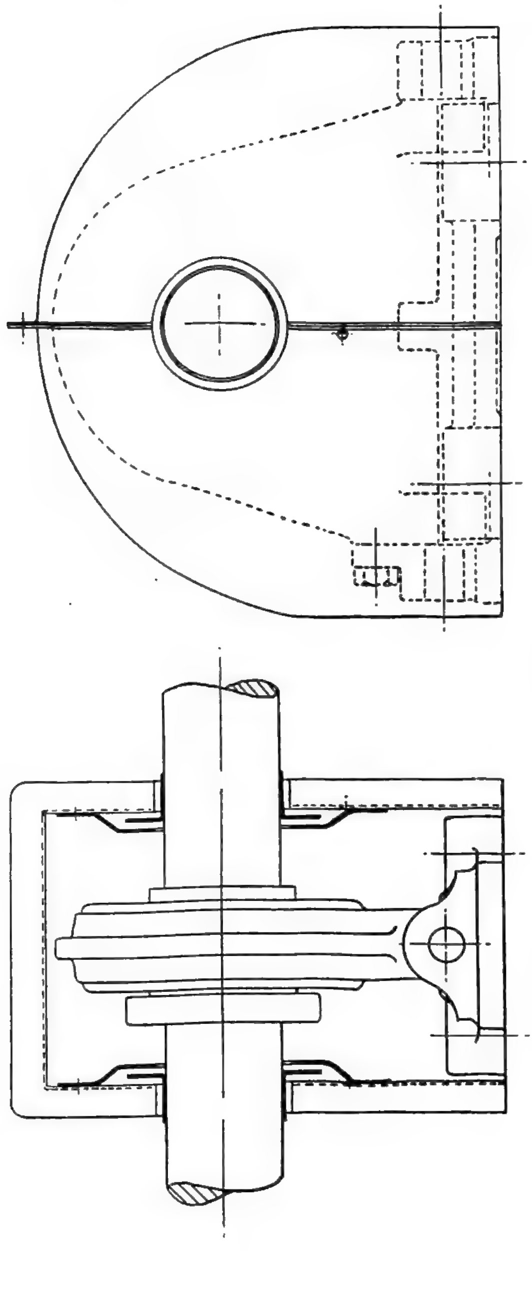


Fig. 21—Method of Protecting Plummer Block Bearing with Sheet Metal Shroud.

With vertical shaft applications the form of protection is similar to that employed on horizontal shafts, and Fig. 24 illustrates a common type of closure used on vertical moulding spindles, where the ambient conditions are very dusty.

Sometimes it is necessary to prevent the ingress of corrosive vapours to a bearing, especially on vertical mixers, and Fig. 25 shows an effective hydraulic seal for slow speed applications.

When oil is the lubricating medium it is normally a requirement that the closure be effective in retaining it in the bearing housing, and Fig. 26 illustrates two types of closures which have proved their efficiency in preventing escapage of oil. (*a*) is a modification of the standard pattern end cover with annular grooves, and in this case a sheet metal thrower overhangs a gutter, formed in the boss of the end cover. This gutter prevents any leakage of oil flung from the rotating thrower and returns it to the working parts of the bearing. (*b*) is perhaps to be preferred as it obviates the necessity of providing a sheet metal thrower, which may be expensive to produce in small quantities. The distance piece, interposed between the inner race of the bearing and the shaft shoulder, has two grooves machined in its outside diameter with an intervening knife edge. Any oil that is flung off the inner end of the distance piece will be trapped by the gutter and thus drain back in to the housing. Should any escape past the gutter it will be thrown off the knife edge into the annular chamber and drain back, through the orifice provided, into the housing. It will be appreciated that it is not possible, in a pamphlet of this nature, to deal with all types of closures as naturally there are many variations of those already dealt with However, the bearing manufacturers are always only too pleased to advise as to the most efficient and economical form of protective devices to employ for any specific application, on receipt of complete details of the conditions appertaining.

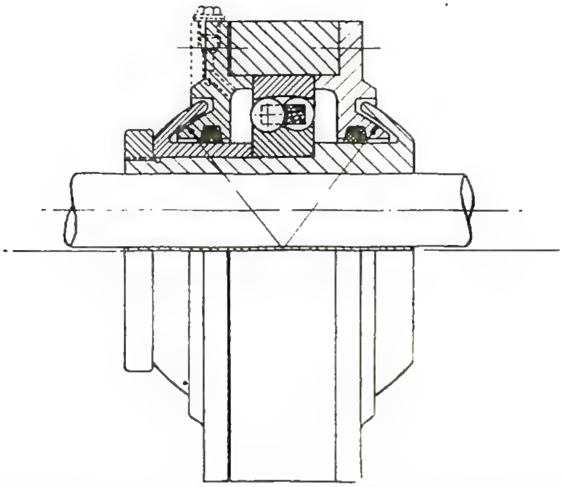


Fig. 22—Method of Protecting Self-Aligning Ball Bearing.

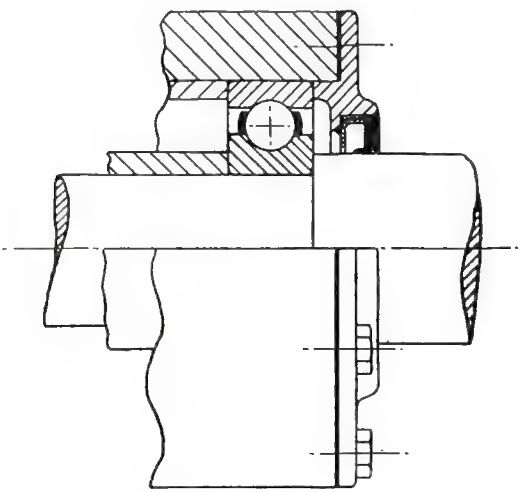


Fig. 23—Application of Proprietary Type Seal.

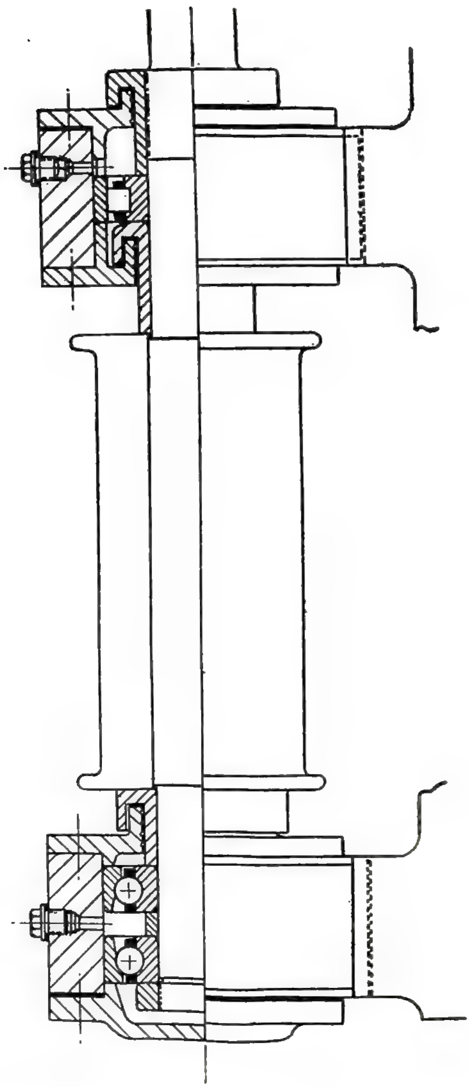


Fig. 24—Method of Protecting Bearings on Vertical Moulding Spindle.

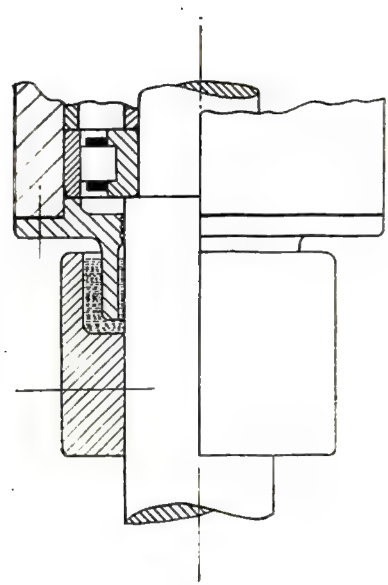


Fig. 25—Hydraulic Seal for Vertical Shaft Application.

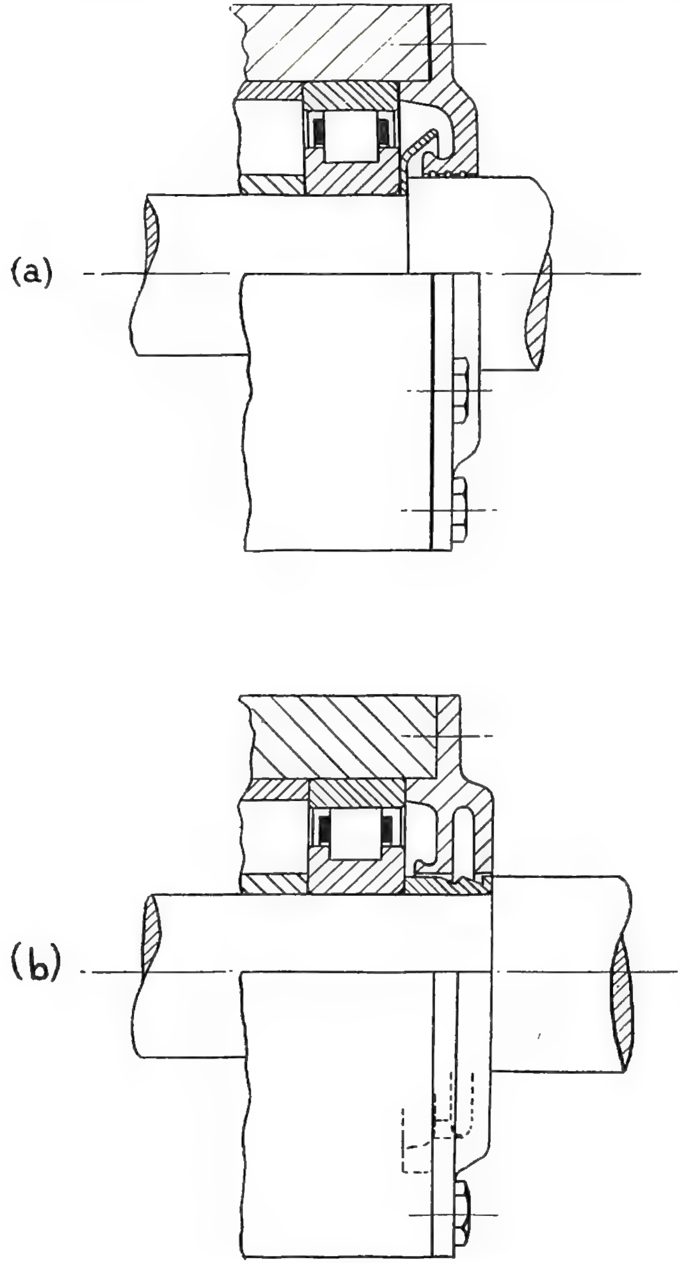


Fig. 26—Closures for Retaining Oil.

LUBRICATION.

It is equally important for ball and roller bearings to be efficiently lubricated as it is for them to be efficiently protected and the results obtained will depend largely on the attention paid to lubrication, both during assembly of the bearings and subsequently during service.

In the majority of applications a correctly lubricated bearing will require only a small replenishment over a long period.

As previously stated grease lubrication is permissible in most applications employing ball and roller bearings and a good grease should have a mineral soap base such as lime, soda or lithium. Such greases should, however, contain no free mineral acid and should be free from alkali or foreign matter.

It is also advisable to avoid the use of filling agents such as talcum, although additives in the form of graphite and zinc oxide are sometimes used. When graphite is used it must be in the colloidal state and free from ash in order to eliminate the danger of lapping of the running tracks and rolling elements.

An important feature in a bearing grease is one of stability; also the grease should have no tendency to gum, thin out or separate either in service or during periods when the machine or apparatus, to which the bearings are fitted, is idle.

The most economical grease to use is one having a lime soap base and such grease is suitable where the speed is moderate and the running temperature does not exceed 100°F. It should be of medium consistency and have a melting point of approximately 200°F.

Where the speed is high for the size of bearing involved and/or where the temperature exceeds 100°F. but does not exceed 200°F., then a grease having a soda soap base should be used. The melting point of these greases is in the region of 300°F. to 350°F., the consistency being correspondingly higher. Care must be exercised in selecting a soda soap grease of the correct consistency for the duty involved as the softer grades are suitable for running temperatures up to 150°F., whilst the stiffer grades, known usually as high melting point greases, will give satisfactory service at temperatures up to 200°F.

More expensive greases, which are becoming increasingly popular, are those having a lithium base as some of these have proved to be satisfactory over a wide range of temperature conditions at speeds higher than those normally attained with grease lubrication.

Users often request information concerning the maximum speeds at which bearings will run with grease lubrication but this depends upon the size and type of bearing involved. It can therefore only be determined with any degree of accuracy by results obtained under actual running conditions. However, the well-known firms of bearing manufacturers publish some useful information on this subject but when in doubt the manufacturer should always be consulted.

Before fitting, the bearings themselves should be packed with the correct grade of running grease, care being taken to ensure that it is well pressed into the working parts, especially between the outside diameter of the inner race and the bore of the cage which retains the rolling elements. When carrying out this operation the importance of keeping the grease clean cannot be too highly stressed as if a few particles of foreign matter are allowed to mix with the grease the life of the bearing will be much shorter than it would have been otherwise.

The spaces in the housings on each side of the bearing should be lightly filled with grease of the same grade so that when the components are fitted together the grease will be maintained in intimate contact with the sides of the bearing, to provide a limited storage of lubricant.

Care must be taken to avoid cramming the end covers and housing as sufficient volume must be left for the excess grease which will be forced out of the bearing during its initial run.

The frequency with which replenishments are necessary depends largely upon the actual operating conditions. With applications running continuously 24 hours per day a small charge once a month or even shorter intervals may be necessary. For applications running under normal duty of load and speed a small replenishment every three to six months is usually sufficient and on relatively slow speed applications, such as lineshafting, regreasing is usually carried out once every twelve months.

Care must be taken not to overcharge with grease and when replenishments are made, the aim should be to introduce just sufficient to disturb the existing grease so as to maintain it in contact with the sides of the bearing. The importance of avoiding overcharging cannot be too highly stressed as a tightly packed housing will result in grease churning with consequent overheating.

When it is necessary to employ oil lubrication, because of minimum frictional requirements, high temperature or high speed, a good quality mineral oil should be used, the viscosity of which should be selected to suit the working conditions.

It is advisable to avoid the use of vegetable and animal oils as these are likely to become rancid under some conditions and have an adverse effect on the bearing working surfaces.

For applications where the frictional resistance must be kept to a minimum a light machine spindle oil should be used, one having a viscosity in the region of 52 Redwood seconds at 100°F, is usually satisfactory. Such an oil, having a low viscosity, will tend to quickly drain away from the bearing working surfaces when the parts are stationary and therefore the higher the viscosity the less the risk of corrosion due to the surfaces being left exposed.

For high speed conditions an oil of light to medium viscosity should be employed and those which are usually found to be satisfactory have a viscosity in the vicinity of 135 Redwood seconds at 100°F.

Where the running temperatures exceeds 200°F. it is usually advisable to use a steam cylinder oil although an oil of lower viscosity may be suitable at temperatures not greatly in excess of 200°F.

In the majority of applications, where the shafts lie in the horizontal plane, oil wells are provided at the bearing positions and these are generally quite efficient provided the oil is maintained approximately half-way up the lowest ball or roller in the bearing.

In automobile gearboxes and industrial gearboxes where the gears are lubricated with oil, it is common practice to allow the bearings to be lubricated by the oil splash or mist which is present inside the gear casing.

This is quite a satisfactory method provided the oil is changed periodically to ensure that any foreign matter, such as metallic particles, are not allowed to accumulate and gain access to the bearings.

In some applications it may be necessary to lubricate the bearings by means of oil sprays, whilst in others a drip feed, which can be regulated from a few drops per minute to a steady flow, may be used, the actual method to adopt depending upon the working conditions.

Drip feed lubricators are very often applied to vertical shafts where it is not always practicable to retain the oil in close proximity to the bearings.

In view of the modern tendency for heavy loading at high speeds the temperature generated by ball and roller bearings sometimes presents another problem and in such cases it is common practice to pump a copious supply of oil to the bearings. This

supply is more than would be necessary for actual lubrication purposes and is to provide a cooling medium, the oil flow being continuous, and the lubricant is cooled before re-entering the bearing housing.

For lightly loaded applications, such as internal grinding spindles, where the speed is extremely high, oil-mist lubrication is used with satisfying results. With this method the oil is injected with air and is blown through the bearings in the form of a mist or a very fine spray. Only a small quantity of oil is required and an additional advantage is that the air flow keeps the bearings cool and also assists in preventing grinding dust, etc., from entering the housing. It is important to ensure that the air supply is free from moisture and, together with the oil supply, must be turned on before the machine is started, as with this system no reserve of oil remains in the housing.

CAUSES OF FAILURE.

Provided a ball or roller bearing is accurately fitted, correctly protected, adequately lubricated and loaded within its calculated capacity then there is no reason, apart from a defective bearing, why it should not give lasting and satisfactory service. In fact in the majority of applications the bearings will last as long as the machine in which they are installed.

In all fairness to the bearing manufacturers it must be stated that the number of breakdowns due to defective bearings is extremely small and in most cases they are due to the failure on the part of the user or the manufacturer of the machine to attend to the elementary rules laid down in the preceding pages of this pamphlet.

Briefly, failures may be classified as under :—

(i) Damage during fitting.

(ii) Mounting errors.

(iii) Incorrect lubrication.

(iv) Inadequate protection.

Whenever bearing failure occurs the troublesome bearing should be returned to the manufacturer as early as possible as co-operation with him, in the early stages, ensures that any necessary rectifications to the application may be carried out before the damage reaches major proportions.

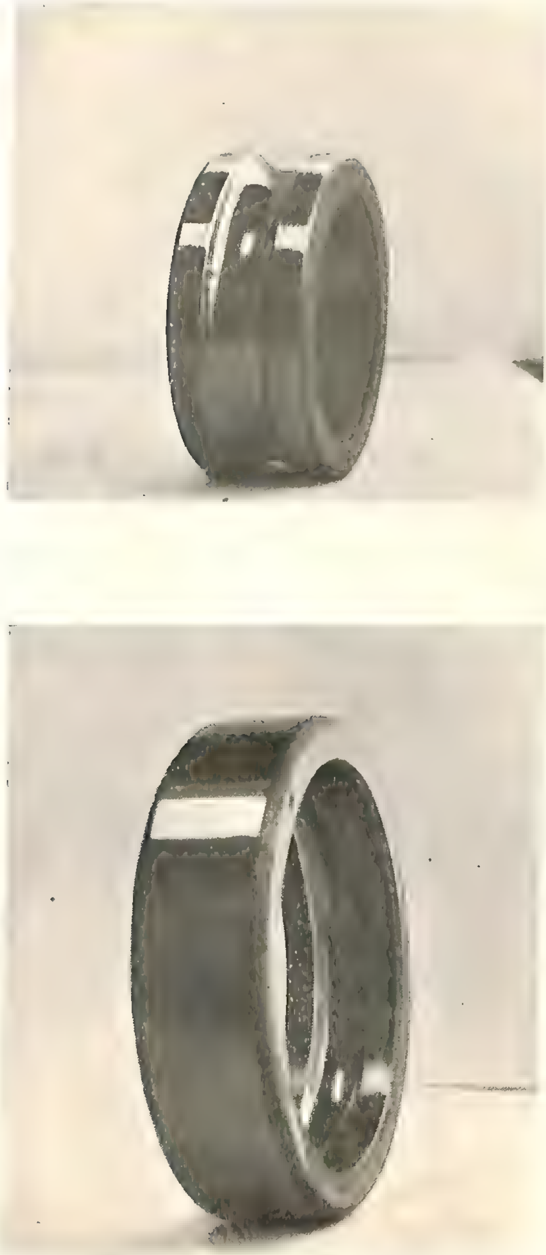


Fig. 27—Damage Due to Incorrect Assembly.

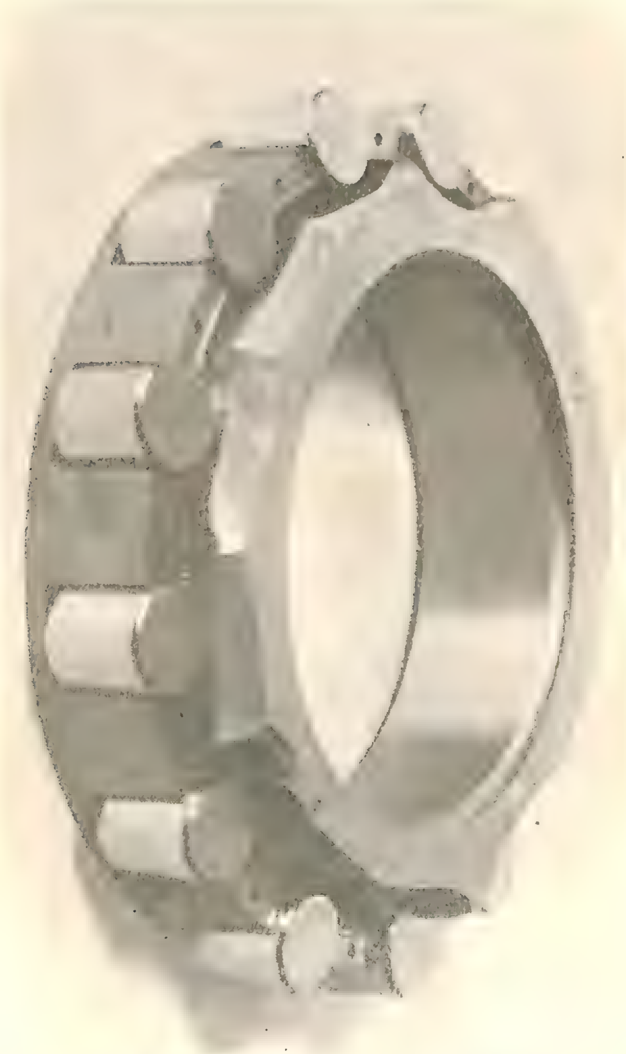


Fig. 28—Damage Due to Bad Fitting.

The best way to appreciate the cause of a bearing failure is to examine some of the more common types of breakdowns that occur and the following illustrations have been included in the hope that they will serve as a warning to future users and that such failures will tend to decrease. It must be emphasised that these photographs have been specially selected to illustrate specific cases of failures that have been due to definite causes, and usually the trouble is of a more complex nature, as one trouble leads to another if not detected in the early stages of breakdown.

Fig. 27 illustrates how damage can be inflicted on a ball bearing as a result of mounting one race by means of its companion race. In this case heavy blows were imposed on the outer race in order to force the inner race on to its seating, with the result that the balls indented the edges of the running tracks. The indentations caused by the balls can be clearly seen on the illustration and, of course, the balls would be correspondingly damaged. Having suffered these initial defects the bearing would be noisy under running conditions, and if the damage was sufficiently severe early failure would occur. Avoidance of these indentations can easily be ensured by applying direct pressure only to the race which is being fitted.

Similar damage can be caused to roller bearings which have lips on both the inner and outer races and Fig. 28 shows an inner race of such a bearing. In this instance the race lip actually broke away at positions corresponding to the roller spacings, a sure indication of fitting damage.

The cage cap and rivets were removed before the photograph was taken in order to show the damage more clearly.

Another point to remember when mounting roller bearings is the importance of keeping the inner and outer races square with one another during the final positioning of the two races, and Fig. 29 shows how the running track may be scored due to canting or twisting the outer race during mounting, necessitating using force to slide it over the rollers.

Failure to ensure that the abutment shoulders are true and square with the axis of rotation accounts for a lot of the trouble experienced, and Fig. 30 shows the outer race of a rigid roller bearing which was canted in the housing. The left-hand illustration shows the part of the running track which supported the working load. It will be noticed that brittling started from the stamped side of the race and gradually spread towards and beyond the centre of the track.

The right hand illustration shows the other half of the running track. In this case the loading was heavier towards the un-

Fig. 29—Damage Due to Canting During Fitting.

Fig. 30—Damage as a Result of a Canted Race.

stamped face of the outer race, so it will be appreciated that the heavy loading moved from one side of the race to the other ; a positive proof that the race had been canted relative to its inner race. Fretting is also apparent on the outside diameter of this race, on the right hand illustration, which corresponds with the position of the heavy loading shown on the left-hand picture.

Fig. 31 shows what may be produced by making the races of a bearing too tight a fit on their seatings. In this case a ball bearing was involved and the photograph shows how the inner track brittled deeply during the initial running stages and then continued in service without further breakdown. This was due to the elimination of diametral clearance in the bearing as a result of excessive interference between the races and their seatings, which produced a heavy preload between the balls and the running tracks. Once the tracks had brittled sufficiently the preload was released and the balls tended to reform a smooth path, but, of course, the damage had been sufficiently severe to result in a noisy bearing.

A bearing that has been incorrectly lubricated usually shows evidence of cage wear but it should be mentioned that canting can also produce wear of the cage.

Fig. 32 illustrates an inner race and the halves of a riveted pattern cage which have suffered from defective lubrication. The cage has worn severely in the bore where it was carried on the outside diameter of the inner race, the wear being more severe at one side, so much so that it has exposed a rivet hole at one position.

Ridges of dried grease can be seen on each side of the running track which indicates that the lubricating grease used was ineffective and a better quality should have been employed.

Fig. 33 shows a disassembled roller bearing that was inadequately protected, with the result that moisture gained access to the various parts causing severe rusting, which is clearly illustrated. This rusting has resulted in the breakdown of the running tracks and roller surfaces with consequent wear and reduction in the load carrying capacity.

CONCLUSION.

The title of this pamphlet commences : "An Introduction to . . . ." and that is all it is intended to be. It does not pretend to be exhaustive as many more pages would have to be written before the subject of ball and roller bearings could be covered in detail. However, it is hoped that those readers who are directly engaged with the design of apparatus employing ball and roller bearings

will have derived sufficient benefit to enable them to identify the type of bearing necessary for the duty involved ; also to be able to apply the elementary principles of mounting necessary for trouble free service.

Finally, the author acknowledges his grateful thanks to the Hoffmann Manufacturing Co. for their kind permission to publish this pamphlet.

Fig. 31—Damage Due to Tight Fitting.

Fig. 32—Damage Due to Defective Lubrication.

Fig. 33—Damage Due to Inadequate Protection.

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3. Deflection of Shafts and Beams (Instruction Sheet). } Connected.
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5. Horse-Power of Leather Belts, etc.
6. Automobile Brakes (Axle Brakes).
7. Automobile Brakes (Transmission Brakes). } Connected.
8. Capacities of Bucket Elevators.
9. Valley Angle Chart for Hoppers and Chutes.
10. Shafts up to 5½ inch diameter, subjected to Twisting and Combined Bending and Twisting.
11. Shafts, 5¾ to 26 inch diameter, subjected to Twisting and Combined Bending and Twisting.
12. Ship Derrick Booms.
13. Spiral Springs (Diameter of Round or Square Wire).
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16. ,, ,, (Plate Clutches).
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19. Internal Expanding Brakes. Angular Proportions for Self-Balancing. } Connected.
20. Referred Mean Pressure Cut-Off, etc.
21. Particulars for Balata Belt Drives.
22. ⅞" Square Duralumin Tubes as Struts.
23. 1" ,, ,, ,, ,,
24. ¾" Square Steel Tubes as Struts (30 ton yield).
25. ⅞" ,, ,, ,, (30 ton yield).
26. 1" ,, ,, ,, (30 ton yield).
27. ¾" ,, ,, ,, (40 ton yield).
28. ⅞" ,, ,, ,, (40 ton yield).
29. 1" ,, ,, ,, (40 ton yield).
30. Moments of Inertia of Built-up Sections (Tables).
31. Moments of Inertia of Built-up Sections (Instructions and Examples). } Connected.
32. Reinforced Concrete Slabs (Line Chart).
33. Reinforced Concrete Slabs (Instructions and Examples). } Connected.
34. Capacity and Speed Chart for Troughed Band Conveyors.
35. Screw Propeller Design (Sheet 1, Diameter Chart).
36. ,, ,, ,, (Sheet 2, Pitch Chart).
37. ,, ,, ,, (Sheet 3, Notes & Examples). } Connected.
38. Open Coil Conical Springs.
39. Close Coil Conical Springs.
40. Trajectory Described by Belt Conveyors (Revised, 1949).
41. Metric Equivalents.
42. Useful Conversion Factors.
43. Torsion of Non-Circular Shafts.
44. Railway Vehicles on Curves.
46. Coned Plate Development.
47. Solution of Triangles (Sheet 1, Right Angles).
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49. Relation between Length, Linear Movement and Angular Movement of Lever (Diagram and Notes).
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